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DESIGN CONSIDERATIONS FOR HIGH EFFICIENCY WET HELIUM EXPANSION ENGINE

PREPARED UNDER FERMILAB SUBCONTRACT NO. 95072
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FOR

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C O N T E N T S

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1. INTRODUCTION:

A new concept in wet helium expansion engines was originally described in CCI Report No. 390-113.[1] A feasibility study of the mechanical aspect for such an engine was made, and a proposed mechanical drive and sequencing control system was developed and reported in CCI Report No. 390-114.[2]

Combining two engines to run in parallel in order to reduce the effects of the inlet and discharge pulses caused additional mechanical problems. It was now necessary to set up and maintain synchronous operation of two engines. The method selected was to drive both engines from a common electric motor and gear reducer. A common output shaft is indexed to right-angle drive gear boxes which are indexed to the main reverser mechanism. Once this mechanism is set up, there is no way, barring breakage, that the units can become unsynchronized.

The report deals with sizing the drive motor, the soft couplings, solenoid valves, and pressure balancing. A cost analysis is also presented.

2. POWER REQUIREMENTS:

The basic power requirement of this engine is very low because of the balanced forces on the piston. Theoretically, the only power required is to overcome gear box and friction losses. However, it is possible for the forces to become unbalanced due to valve delay, flow characteristics, different chamber volumes, shifting controller set points, etc. As long as the resultant force does not overload the rollnut, the engine should continue to operate without stalling.

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[1]Vander Arend, P.: Wet Expansion Engine for Helium Refrigeration Application. 6 September 1979, CCI Report No. 390-113.

[2]Harmes, C.: Mechanical Drive and Control System for High Efficiency Wet Helium Expansion Engine. 18 September 1979, CCI Report No. 390-114.

The motor horsepower was selected to provide sufficient torque to continue driving the pistons even with the maximum allowable unbalance.

2.1 Calculations:

The intake, expansion, and exhaust strokes occur within an 8 second time span. Since the reverser mechanism shaft has a 2:1 reversing pitch, the intake and expansion stroke requires 2/3 x 8 = 5.33 seconds. The exhaust stroke requires 2.67 seconds.

Motor Speed x Reduction x Pitch x 5.33 sec = 8-3/4 in.

$$\text{Motor Speed} = \frac{8.75 \times 30}{2 \times 5.33} = 24.625 \text{ RPS}$$

$$= 1{,}477 \text{ RPM}$$

$$\text{Reverser Speed} = \frac{1477}{30} = 49.25 \text{ RPM}$$

Maximum allowable axial load on reverser is 505 lbs.

$$\text{Maximum Torque} = \frac{PL}{2\pi\rho_1}$$

$$= \frac{505 \times 2}{2\pi \times 50\%}$$

$$= 321 \text{ in.lbs (Intake \& Expansion Strokes)}$$

$$\text{Maximum Torque} = \frac{505 \times 4}{2\pi \times 50\%}$$

$$= 642 \text{ in.lbs (Exhaust Stroke)}$$

$$\text{HP} = \frac{\text{Torque} \times \text{RPM}}{63{,}025}$$

$$= \frac{321 \times 49.25}{63{,}025}$$

$$= 0.25 \text{ HP (Intake \& Expansion Strokes)}$$

$$\text{HP} = \frac{642 \times 49.25}{63{,}025}$$

$$= 0.50 \text{ (Discharge Stroke)}$$

Maximum HP theoretically consumed is when one engine is intaking while the other is discharging.

$$HP = 0.25 + 0.5 = 0.75$$

$$\text{Motor HP} = 0.75 \times \frac{1750}{1477} = 0.89$$

Allow 60% efficiency of gear boxes.

Motor Req'd. = 1.48 HP = 1-1/2 HP

Torque Req'd. = 321 + 642 = 963 in.lbs

Gear Box 1120 in.lbs. Rating

Right-angle Drive 1044 in. lbs Rating

Allowable ΔP across piston:

$$\Delta P = \frac{505\#}{4^2 \times \frac{\pi}{4}} = 40 \text{ psi}$$

The 1-1/2 HP motor was selected on the basis that should a 40 psi pressure differential exist across the piston, the motor would continue driving the engine. A pressure differential greater than 40 psi causes a piston shaft force which exceeds the rated load of the reverser mechanism, and the engine should be stopped.

3. PRESSURE BALANCING:

The concept of a pressure balanced piston has two very significant aspects. First, the opposed pressures on the piston ends create a zero net force on the piston rod. Theoretically, to move the piston only friction forces need be overcome; but, practically, some net force due to small pressure differentials will exist. Secondly, the balanced pressures will reduce leakage by the piston from the cold end to the warm end or vice versa.

The pressures will be monitored, and balancing will be accomplished by means of a pressure differential controller and throttling valve.

4. SOLENOID VALVES:

The basic schematic illustrating the valve arrangement is shown in Figure 1:

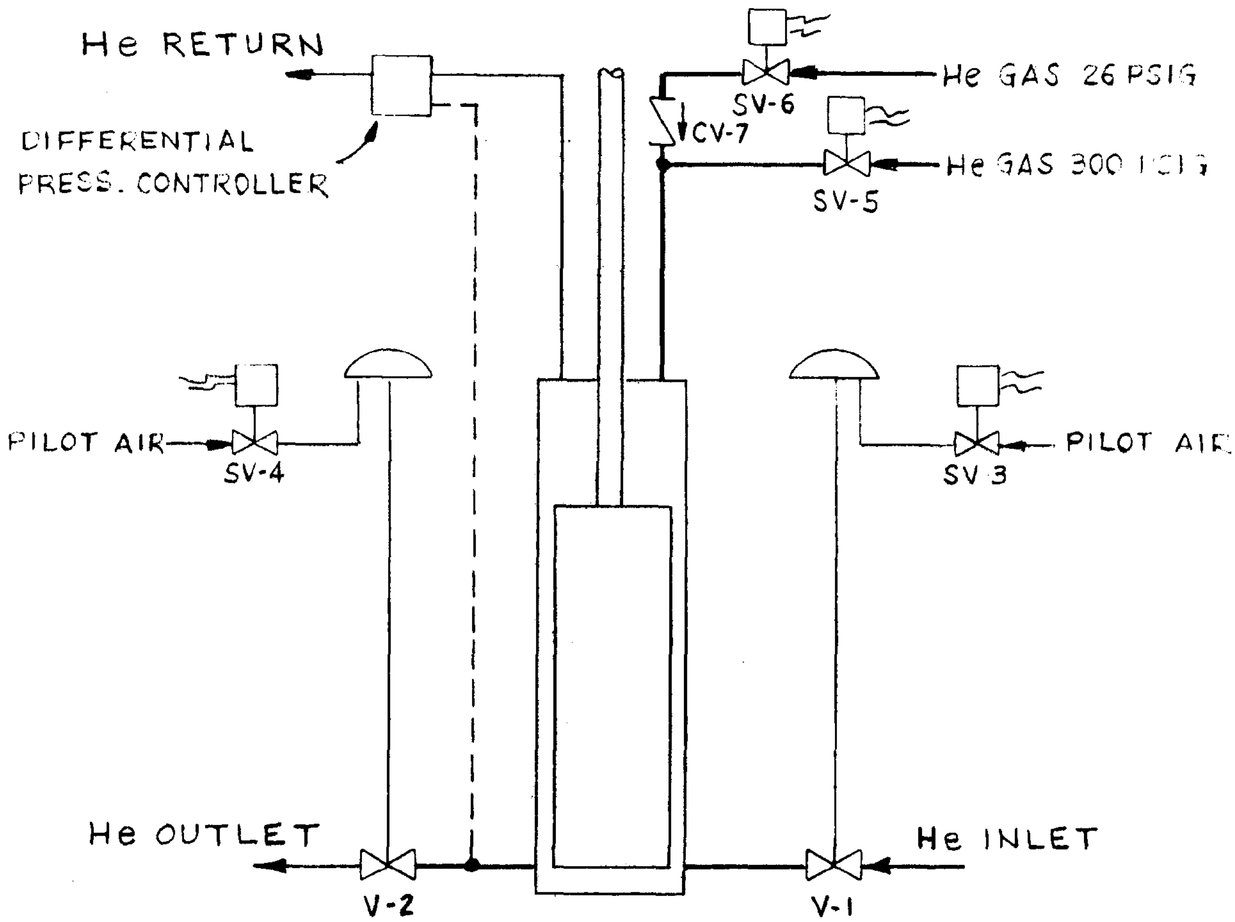

FIGURE 1

It is assumed that V1 and V2 are air-operated diaphragm valves. The response will be slightly slower than a direct operating electric solenoid valve, but an analysis of the full operating cycle shows that this is of no consequence.

4.1 Bottom Dead Center (BDC or 0°):

4.1.1 Pilot valve SV3 opens to emit air into the diaphragm of valve V1 causing V1 to open.

4.1.2 Solenoid valve SV5 is energized to emit helium gas to top volume of cylinder.

4.1.3 Valves V2, SV4, and SV6 are all closed.

4.1.4 During this time, a slight dwell cut into the reverser screw provides a short duration when the piston does not move. A slight delay in valve response will not do any harm.

4.2 5°:

4.2.1 Valve SV5 closed.

4.3 Expansion Stroke (To Approximately 155°):

4.3.1 The differential pressure controller maintains a pressure balance across the piston.

4.4 155°:

4.4.1 Pilot valve SV3 closes to close V1.

4.4.2 Any delay or response time can be compensated by a change in the electronic countersetting.

4.5 Continuation of Expansion Stroke (To 180°):

4.5.1 The helium at the head end of the cylinder is allowed to expand.

4.5.2 Excess pressure at rod end of cylinder bleeds out of the pressure differential controller.

4.6 180°:

4.6.1 The helium gas has expanded to 26 psig.

4.6.2 Pilot valve SV4 opens to open valve V2.

4.6.3 Valve SV6 opens to emit helium gas at 26 psig to rod end of cylinder.

4.6.4 A dwell cut into reverser mechanism compensates for the response time of the valves.

4.7 Discharge Stroke (180 to 360°):

4.7.1 Expanded helium gas is evacuated through V2.

4.8 End of Discharge Stroke (360°):

4.8.1 Pilot valve SV4 closes to close valve V2.

4.8.2 Valve SV6 closes.

The piston speed during the expansion stroke is:

$$\frac{8\text{-}3/4 \text{ in.}}{2/3 \times 8 \text{ sec}} = 1.641 \text{ in./sec (expansion)}$$

The piston speed during the discharge stroke is:

$$\frac{8\text{-}3/4 \text{ in.}}{1/3 \times 8 \text{ sec}} = 3.281 \text{ in./sec (discharge)}$$

Because of the reversing requirement and the dwell cut at the end of the stroke, the piston remains at rest for a period of approximately 0.3 to 0.5 sec. This is ample time to accomplish valve response.

5. SOFT COUPLING:

Rapid opening of the inlet solenoid valves after the completion of the discharge stroke can produce shock loading on the piston in the upward direction. This shock load can be transmitted through rigid linkage to the reverser mechanism and reduce the life of the reverser nut.

The connection between the piston shaft and the reversing carriage has been equipped with shock absorbers to protect the reversing nut. Presently, there are holes to accommodate six "sandwich" type mounts. If prototype tests show that six mounts are too rigid, the number may be decreased or more resilient ones selected.

With a net force on the piston of 505 lbs, the deflection of the shock absorbers is:

$$\text{Deflection} = \frac{\text{Load}}{n \times Kc} = \frac{505}{6 \times 1200 \text{ lb/in.}} = .070 \text{ in.}$$

This deflection is 0.8% of the total travel.

6. COST ESTIMATE:

The cost to manufacture two engines is tabulated below. This cost does not include valves or the cold box, only the engine assemblies.

Item	Description	Qty. Req'd.	Price Each	Total
1	Elect. Motor, 1-1/2 HP DC Shunt Wound V18150A-B	1	325.20 x .85	276.42
2	Speed Controller, Run-Stop 63405VE150	1	519.00 x .85	441.15
3	Speed Reducer, Single Reduction, Boston SF326 30:1	1	334.59 x .85	284.40
4	Right-angle Drive, Heavy Duty, Ratio 1:1, Boston VR146, No Vert. Down Shaft Req'd.	1	328.21 x .85	278.98
5	Same as Item 4, Except Opposite Relative Rotation	1	348.50 x .85	296.23
6	Couplings, Falk #40T20, 1" Bore w/1/4x1/4 Key	4	51.68	206.72
7	Mech. Actuator (MD-129263) 2:1 Ratio,8-/4" Stroke, BR2009-Special	2	2,000.00	4,000.00
8	Ball Bushings, Thomson #A-162536	8	14.61	120.24
9	Flanged Cartridge Unit, FMC #FC-B22420H	4	34.75	139.00
10	Proximity Switch, Newark S/N 33F1208	2	3.75	7.50
11	Magnetic Actuator, Newark S/N 33F1211	2	2.00	4.00
12	Fabrication (MD-129255) Bearing Support & Guide	2	240.00	480.00

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Item	Description	Qty Req'd.	Price Each	Total
13	Fabrication (ME-129252) Reciprocating Arm	2	240.00	480.00
14	Fabrication (MC-129260) Shock Absorber & Shaft Coupling	2	50.00	100.00
15	Fabrication (MC-129267) Shaft (Piston)	2	200.00	400.00
16	Fabrication (MC-129259) Seal & Bearing Housing	2	250.00	500.00
17	Fabrication (MD-129271) Piston	2	400.00	800.00
18	Fabrication (ME-129253 Cylinder	2	400.00	800.00
19	Rotopulser, Dynapar #80D	2	290.00	580.00
20	Counter-Controller Dynapar #544C Special	2	1,128.00	2,256.00
21	Adjustable Rate Multiplier Dynapar	2	180.00	360.00
22	Shaft Seal	2	25.00	50.00
23	Linear Bearing, LM76 #L-1625-18	2	10.13	20.26
24	Shock Absorbers, Lord Kinematics #J4624-45	Set of 12	100.00	100.00
25	Fabrication (MC-129268) Shaft (Main Drive)	1	75.00	75.00
26	Guide Shafts (MC-129262)	4	35.00	140.00
27	Piston Sealing Rings	2 Sets	250.00	500.00
28	Fabrication (ME-129251) Stationary Housing	2	400.00	800.00
			TOTAL COST:	$14,645.90

The estimated costs of the valves are tabulated below:

Valve V1 with Operator	$850.00
Valve V2 with Operator	$850.00
Solenoid Valve SV3	$ 40.00
Solenoid Valve SV4	$ 40.00
Solenoid Valve SV5	$ 40.00
Solenoid Valve SV6	$ 40.00
Check Valve CV7	$ 20.00